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Energy Impact of Residential Ventilation Standards in California

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ABSTRACT

The California Energy Commission is considering upgrading the State energy code, known as Title 24, to require mechanical ventilation based on the requirements of ASHRAE Standard 62.2-2004, Ventilation and Acceptable Indoor Air Quality in Low-Rise Residential Buildings*. These requirements will require mechanical ventilation systems to be installed in virtually all new homes, but allows for a wide variety of design solutions. These solutions, however, may have different energy costs and non-energy benefits. The authors have used a detailed simulation model to evaluate the energy impacts of common and proposed mechanical ventilation approaches for a variety of climates. These results separate the energy needed to ventilate from the energy needed to condition the ventilation air, from the energy needed to distribute and/or temper the ventilation air. The results show that exhaust systems are generally the most energy efficient method of meeting the proposed requirements, but that supply and balanced systems can provide additional non-energy benefits..*

INTRODUCTION

The purpose of ventilation is to provide fresh (or at least outdoor) air for comfort and to ensure healthy indoor air quality by diluting contaminants. Historically, people have ventilated buildings to provide source control for both combustion products and objectionable odors (Sherman (2004). Currently, a wide range of ventilation technologies is available to provide ventilation in dwellings including both mechanical systems and sustainable technologies. Most of the existing housing stock in the U.S. uses infiltration combined with window opening to provide ventilation, sometimes resulting in over-ventilation with subsequent energy loss; sometimes resulting in under-ventilation and poor indoor air quality. Sherman and Dickerhoff (1994), Sherman and Matson (2002) have shown that recent residential construction has created tighter, energy-saving building envelopes that create a potential for under-ventilation. Infiltration rates in these new homes average 3 to 4 times less than rates in existing stock. As a result, new homes often need mechanical ventilation systems to meet current ventilation standards. McWilliams and Sherman (2005) have reviewed such standards and related factors.

As part of its energy code, the State of California has always included minimum ventilation requirements to protect the health and safety of occupants. In previous versions of this code, minimum ventilation in homes could be met by a combination of infiltration and natural ventilation (i.e. window opening). For the next round of that code, due to take effect in 2008, the State is considering mechanical ventilation requirements. This paper reports on the energy impacts of those requirements.

Because of the effects it has on health, comfort, and serviceability, indoor air quality in our homes is becoming of increasing concern to many people. According to the American Lung Association a number of factors within our *homes* have been increasingly recognized as threats to our respiratory health. The Environmental Protection Agency lists poor indoor air quality as the fourth largest environmental threat to our country. Asthma is the leading serious chronic illness of children in the U.S. Construction defect litigation and damage are on the increase in new houses and some of this increase is related to indoor air quality problems such as moisture. Residential ventilation can address many of these indoor air quality problems.

Traditionally residential ventilation was not a major concern because policy makers believed that between operable windows and envelope leakage, people were getting enough outdoor air. In the three decades since the first oil shock, houses have become much more energy efficient. At the same time, the types of materials used in furniture, appliances, and building materials in houses have changed. People have also become more environmentally conscious not only about the resources they were consuming but about the environment in which they lived.

All of these factors have contributed to an increasing level of public concern about residential indoor air quality and ventilation. Where once there was little concern about the residential indoor environment, there is now a desire to define levels of acceptability and performance. Because states and other jurisdictions have a responsibility to protect the health and safety of their populace, more and more of them are considering changes to their codes and regulations.

Ventilation can be an energy consuming activity that provides acceptable indoor air quality. There is no fixed "right" amount of ventilation in the same sense that there is no fixed "right" size of furnace or air conditioner for all houses and climates. To find the minimum requirements for thermal conditioning, one must determine the thermal load and the desired thermal conditions to be met. Similarly, to determine how much ventilation is necessary, one must look at the sources and emissions of concern to indoor air quality (e.g. moisture levels and pollutant loads) and the desired level of indoor air quality. Based on these two factors, the choice of a ventilation level must be made by trading off various costs (e.g., energy costs, first cost, risks) with the benefits associated with the building service (e.g., health and comfort).

The problems of determining appropriate minimum ventilation rates are much more complex than those confronted in efforts such as determining thermal insulation levels. Thermal loads are well studied and can be robustly estimated from internal gains and weather conditions. In contrast, pollutant sources tend to be highly variable among different households and are quite dynamic. Thermal comfort can be predicted quite reliably from just a few environmental parameters (e.g., temperature, air speed, humidity) whereas acceptable indoor air quality depends on a number of environmental and exposure parameters, many without established value ranges or acceptability criteria.

Because of the complexities of IAQ, ventilation standards and guidelines have followed a much more subjective route than thermal standards and guidelines. Extant ventilation requirements are based on evaluations of what has or has not worked in the past and thus incorporate the experience of experts in the field. As the first and only ANSI-certified residential ventilation standard in the country, ASHRAE Standard 62.2-2004 serves as the starting point for the work in this paper and is summarized below.

Overview of ASHRAE Standard 62.2-2004

In developing this standard ASHRAE recognized that there were many different kinds of houses, many different climates, and many different construction methods. To accommodate these differences, the major requirements were designed with several alternate paths to allow users flexibility. Some requirements are performance based, with specific prescriptive alternatives. The standard recognizes that there are several different ways to achieve a specified ventilation rate and allows both mechanical and natural methods.

There are three main primary sets of requirements in the standard and a host of secondary ones. The three primary sets involve whole-house ventilation, local exhaust, and source control. Whole-house ventilation is intended to dilute the unavoidable contaminant emissions from people, from materials and from background processes. Local exhaust is intended to remove contaminants from specific rooms in which sources are expected to be produced by design (primarily kitchens and bathrooms). Other source control measures are included to deal with those sources that can reasonably be anticipated and dealt with.

The secondary requirements focus on properties of specific items that are needed to achieve the main objectives of the standard. Examples of this include sound and flow ratings for fans and labeling requirements. Some of the secondary requirements as well as the guidance in the appendices help keep the design of the building as a system from failing because ventilation systems were installed. For example, ventilation systems that excessively push moist air into the building envelope can lead to material damage unless the design of the envelope is moisture tolerant.

ASHRAE Standard 62.2 (ASHRAE (2004)) has requirements for whole-house ventilation, local exhaust ventilation, source control and system requirements. In brief they can be summarized as follows:

The whole-house ventilation rate is 3 cfm/100 sq. ft. plus 7.5 cfm/person. It is assumed that 2 cfm/100 sq. ft. can be supplied through infiltration. For most of the country the difference must be supplied by mechanical ventilation, but for most of California operable windows may be used.

Local mechanical exhaust is required in kitchens and bathrooms. Kitchens must have the capacity to exhaust at least 100 cfm through a range hood or provide 5 kitchen air changes per hour. Bathrooms must have the capacity to exhaust 50 cfm or have 20 cfm of exhaust continuously.

For source control: dryers must be vented to outdoors; naturally aspirated combustion appliances may not be inside under some conditions; filtration is required on air handling systems; leaky ductwork is not permitted in garages.

Air moving equipment must be rated and certified to meet its intended use and meet certain noise and airflow specifications.

California Energy and Ventilation Concerns

California's Energy Code, known as Title 24 (Part 6), focuses on cost-effective ways to minimize the energy-related impacts of providing building services such as thermal conditioning, lighting and water heating. Ventilation is an energy consuming activity and considered an energy end-use. Providing acceptable indoor air quality is the building service provided by ventilation.

The purpose of ventilation is to provide acceptable indoor air quality and to enhance durability of systems and materials. This study is part of a project to provide input to the California Energy Commission and its efforts to update the State's energy code: 2005 Building Energy Efficiency Standards for Residential and Non-Residential buildings (referred to as Title 24). McWilliams and Sherman (2005) have examined the relationship of ventilation to health by using ASHRAE/ANSI 62.2-2004 and how this differs from current Title 24 requirements and other relevant codes. A key factor for this project is the effect that any changes to the ventilation requirements in Title 24 have on energy use algorithms and compliance tools already in use by the California building industry. Addressing this issue requires interactions with software developers who provide compliance tools for the California Building Energy Efficiency Standards.

The building ventilation requirements in the current Title 24 standard are primarily engineering based and, as a result, technical feasibility is likely to remain a key driver for these standards. Therefore this project will focus on existing ventilation strategies and technologies.

The following engineering issues related to providing acceptable indoor air quality without large energy penalties were examined :

The usability of occupant control, particularly window opening

- Distribution and mixing of fresh air throughout a house
- The role of unusual sources and/or source control
- The role of air cleaning and particle filtration
- Effects of poor outdoor air quality (e.g., particulates near busy roads or in rural communities)

The current version of Title 24 has ventilation requirements that go beyond what most states require. Other states including Florida, Minnesota and Washington have also adopted minimum ventilation requirements. Other countries, such as Canada, France, Sweden, the United Kingdom and Denmark have specific ventilation requirements. In order to evaluate changes to Title 24, it is important to look at what other codes have specified, as well as what are considered best professional practice (ASHRAE 62.2).

Overview of Residential Ventilation Requirements in Title 24

Mechanical ventilation is required by Title 24 only if the building has a Specific Leakage Area (SLA) less than 3.0, which corresponds (in typical California climates) to approximately 0.26 air changes per hour that would be provided by infiltration in such a house. Of course, the actual ventilation provided by infiltration would vary with seasonal temperature differences and local wind conditions. When mechanical ventilation is specified, Title 24 requires that a whole house ventilation system be installed with the capacity to provide 0.047 cfm/square foot (0.24 L/s/m^2).

If the SLA is less than 1.5, Title 24 states that supply ventilation must be provided with enough capacity to maintain a house pressure greater than -0.02 in. water (-5 Pascals) relative to outside when the other exhaust fans are running. This is to avoid potential backdraft problems with combustion appliances, particularly fireplaces and also to provide some protection in case of future installation of combustion appliances. The standard also requires an air inlet and glass doors for fireplaces, wood, pellet and gas stoves.

When performing energy calculations, Title 24 also assumes that windows will be opened by the occupants whenever the ventilation rate drops below 0.35 air changes per hour.

Ventilation appears in Title 24 not only as a minimum ventilation requirement but also in the form of ventilative cooling. The latter is in the form of natural ventilation, whereby Title 24 assumes that under certain circumstances occupants will open their windows to maximize free cooling. This ventilative cooling servers both purposes of reducing air conditioning and, incidentally, of exhausting indoor contaminants

Next Generation of Residential Ventilation in Title 24

Several changes have been recommended for Title 24 and form the basis for our simulations. These recommendations can be grouped into those for IAQ-related requirements, those for energy-related requirements and those needing more research.

IAQ-Related Requirements:

- ASHRAE Standard 62.2-2004 should be adopted by reference except where overridden by any of the other recommendations of this study.
- Air leakage and window operation shall be deemed to provide the default infiltration of 62.2 and also meet the window opening requirements of the standard, but may not be used to otherwise meet the whole-building ventilation or local exhaust provisions of the standard.
- The mechanical ventilation rates of Section 4.1 of the standard shall be increased by 25 cfm.
- When ducts pass through buffer zones (e.g. a garage, attic or crawlspace) the total leakage shall be limited to 5% of the air handler flow.

Energy-Related Requirements:

- SLA=4 shall be the default value for use in energy calculations, but measured air tightness may be used if known. Current restrictions on tightness below SLA=1.5 should be removed, but this does not impact the simulations.
- Designs with suitable control systems (e.g. a programmable timer) may take energy credit for shutting off the ventilation system up to 4 hours per day.

EVALUATION APPROACH

In order to evaluate the energy impacts of the proposed requirements, an extensive simulation protocol was executed using the REGCAP simulation tool. This tool performs minute-by-minute ventilation, heat and moisture calculations that allow for the dynamic performance of buildings and HVAC components. The small timesteps are computationally and analytically intensive but allow for direct simulation of temporally complex ventilation controls. REGCAP combines a mass balance for air flows with a thermal model including the HVAC system and a moisture transport model. The air flow model allows individual (such as passive vents or flues) and distributed leaks (such as over a wall) to be placed on the building envelope. A rectangular floor plan was assumed and the envelope leaks a were separated into the leaks in each of the four walls. Similarly, each face of the building had floor level and eave height leakage. The attic had leakage in the gable ends and two pitched roof surfaces together with eave height soffits, gable vents, ridge vents and vents in the pitched roof surfaces. The air flow mass balance was performed for two zones: the house and the attic. The attic is particularly important in these simulations because the duct system for heating and cooling is located in the attic and heat transfer and air flows in and out of the ducts are key components of the building load and ventilation air flow network. The flow through each leak was determined by the air flow characteristics of the leak (flow coefficient and pressure exponent) and the pressure across the leak. The pressure across the leak depended on both wind pressures and buoyancy pressures due to indoor-outdoor temperature differences. The wind pressures were separately determined for each individual leak location. The mechanical ventilation systems were integrated into the mass balance as constant flow devices. REGCAP has the ability to use fan curves but they were not used in this study because previous work has shown that the small pressured typical of building envelopes do not significantly change the air flow through the fans. In addition, the specific fan performance curves for ventilation fans are unknown. For the HVAC system , when the system blower is off, the duct leaks in the attic are treated the same way as building envelope leakage (in this case the same as ceiling leakage). With the blower on, the air flow through supply and return grilles was included for the house and air flow through supply and return leaks was included for the attic space. The duct leakage rates were fixed at 2.5% of air handler flow for both supply and return ducts for a total of 5% duct leakage. These ducts are therefore tight enough to satisfy the tight duct credit requirements in Title 24. The blower, duct leak and register flows were all treated as fixed air flow rates that depended on the HVAC operating mode: heating cooling or ventilating. The mass balance for the house and attic was solved by adjusting the internal pressures of the two zones. Although REGCAP can include air flow through open doors and windows (including two-way and interfacial mixing flows) this study did not use this capability. The two zones interacted via the air flow through the ceiling (and duct leaks with the HVAC system blower off). The mass flows through all the individual flow paths were combined. For both the attic and the house an internal pressure shift relative to outside is the only unknown to be determined. Because the equations are highly non-linear, a simple pressure bisection technique was used to determine the attic and house interior pressures as this has proven to be an extremely robust solution technique. More details of the mass balance model can be found in Walker et al. (2005).

All the mass flows depended on the density of the air flow. The temperatures of the various air flows were found from either weather data (for outside air) or the thermal model in REGCAP. The thermal model used:

- overall UA values for the building envelope to determine heat transfer through the envelope of the house. House insulation levels and window performance were based on California Energy Code requirements (California Energy Commission 2005) and included degradation due to incorrect installation per the code requirements.
- solar gain through windows that depends on the solar heat gain coefficient and orientation. The solar part of the model used standardized calculations based on those in ASHRAE Fundamentals (ASHRAE 2005) together with measured solar radiation in the weather data.
- material thermal properties for the attic envelope including thermal conductivity, thermal mass and surface properties for radiation. The latter was particularly important for attics whose temperatures depend strongly on solar radiation.
- the mass flows derived by the air flow mass balance. This includes both air flows through the attic and house envelope together with duct leaks.
- the heat input or removed by the HVAC equipment including latent removal by air conditioning. The output capacity of the furnace for heating assumed the use of gas furnace with 80% AFUE. The furnace blower motor waste heat (85% of blower power consumption) was included as an extra heat source. The furnace blower ran for one additional minute at the end of each cycle with the burners off to purge the system. For cooling, a SEER 13 EER 11 air conditioner was used. The air conditioner capacity calculations depended on air flow across the cooling coil (using the methods in ASHRAE Standard 152 (ASHRAE 2004) fixed at a nominal 400 cfm/ton), the outdoor dry bulb temperature and duct return dry bulb temperature and humidity ratio. The duct return dry bulb temperature was changed from indoor conditions by any heat transfer to or from the return ducts in the attic. The return air temperature and humidity ratio included the air flow into the return for central fan integrated systems. The sensible heat ratio (SHR) depended on the return air humidity ratio using a simple linear correlation based on manufacturer's data. The SHR ramped down from 1.0 to the steady state SHR based on return humidity over a period of three minutes based on the work of Henderson and Rengarahan (1996) and Henderson (1998). The system capacities were determined by analyzing survey results of new California homes (Chitwood 2006) and are summarized in Walker and Sherman (2006). The capacities were generally larger than ACCA manual J and S requirements. In some cases the cooling capacity determines the heating capacity due to the limited packaging alternatives that are commercially available. Primarily this is an issue of furnace blower motor operating ranges that restrict the differences in heating and cooling capacities that can be serviced by an individual blower.
- internal gains were based on California Energy Code requirements (California Energy Commission 2005).

- a radiation heat transfer balance in the attic - primarily to include the effects of solar heating and radiative cooling of the roof deck and how this changes duct and ceiling surface temperatures.

A total of 16 heat transfer nodes were identified including air in the ducts, house, and attic, and used in a lumped heat capacity analysis. At each node the rate of change of thermal energy was equated to the sum of the heat fluxes due to radiation, convection and conduction. This resulted a set of equations that were linear in temperature and were solved simultaneously. This simultaneous solution was found using Gaussian elimination. When the temperatures were calculated they were returned to the airflow model so that new mass flow rates were calculated. This iteration between the thermal and mass flow parts of the simulation was continued until the attic air temperature changed by less than 0.1^{o}C. Because the attic ventilation rates are relatively insensitive to the attic air temperature usually fewer than five iterations between thermal and ventilation models are required.

The moisture model was based on mass balance between outside, the house, the attic, supply and return ducts and a mass storage term in the house. Moisture removal for the air conditioner was based on system total capacity and SHR. Indoor moisture source strengths were taken from draft ASHRAE Standard 160P (ASHRAE 2006) but modified assuming that the house was unoccupied for 8 hours per day (resulting in 2/3 of the generation rate). For the moisture storage a mass transport coefficient and total mass storage capacity were used that were determined empirically by comparing predicted humidity variation to measured field data in houses (from Rudd and Henderson (2006) and Building Science Corporation (2006)). Both the transport and storage terms scale with house size (floor area): the total mass capacity for storage was 12.3 lb/ft^2 (60 kg/m^2) of floor area and the mass transport coefficient was 0.0006lb/(sft^2) (0.003kg/(sm^2)). The resulting damping in indoor air moisture variability is close to the empirical formulation in the Environmental Protection Agency's (2001) Indoor Humidity Assessment Tool that uses a capacitance term to allow only 5-10% of moisture flow to inside to go into the air and assumes the other 90-95% was absorbed by building contents.

The key issue with the use of this particular simulation tool is the ability to account for HVAC system, house and attic air flow, thermal and moisture transport interactions. More details of the simulation tool can be found in Walker and Sherman (2006).

Climates Evaluated

The State of California defines sixteen climate zones for the purposes of its standards. With the exception of the humid climates in the southeast of the United States, these climate zones span most of the climates in the country. In this paper we will focus on only three of the climate zones spanning the range from cold to hot and dry. Results for the other climates are available in Walker and Sherman (2006).

Cold climate: The cold climate is represented by California climate zone 16, or equivalently by IECC climate zone 6 and a representative city might be Chicago or Minneapolis.

Mild climate: The mild climate is represented by California climate zone 3, or equivalently by IECC climate zone 4C and a representative city might be Oakland (Oakland is the city used for California climate zone 3 weather data).

Hot, dry climate. The hot, dry climate is represented by California climate zone 15, or equivalently by IECC climate zone 2A and a representative city might be Phoenix.

Systems Evaluated

The proposed requirements are to have mechanical ventilation that meets ASHRAE Standard 62.2 plus an extra 25 cfm of capacity to allow for periodic turning off of the system (whether as part of a controlled ventilation system or by occupant intervention). The required mechanical ventilation requirements were determined using Equation 1:

$$Q\left(cfm\right) = 0.01A_{floor}\left(ft^2\right) + 7.5\left(N+1\right)$$
$$Q\left(L/s\right) = 0.05A_{floor}\left(m^2\right) + 3.5\left(N+1\right) \quad \textbf{(1)}$$

where N is the number of bedrooms in the house.

For the three house sizes we simulated the air flows are:

1000 ft^2 & 2 bedrooms (3 occupants) $\Rightarrow$ 33 cfm (16 L/s)

1761 ft^2 & 3 bedrooms (4 occupants) $\Rightarrow$ 48 cfm (23 L/s)

4000 ft2 & 5 bedrooms (6 occupants) ⇒ 85 cfm (40 L/s)
Adding the extra 25 cfm (12 L/s) results in:
1000 ft^2 & 2 bedrooms (3 occupants) ⇒ 58 cfm (27 L/s)
1761 ft^2 & 3 bedrooms (4 occupants) ⇒ 73 cfm (34 L/s)
4000 ft2 & 5 bedrooms (6 occupants) ⇒ 110 cfm (52 L/s)

We considered nine different ventilation systems derived from the review of Russell et al (2005) and those used or planned to be used in California. Not all of these systems meet the criteria being proposed for Title 24. Air flow and power requirements were determined from the HVI directory (HVI (2005)). The ones that meet the criteria are as follows:

Continuous Exhaust: A continuous exhaust system is modeled as one of the bathroom fans being replaced by a fan that continuously meets the minimum air flow (and other) requirements. For the 1761 ft^2 (164 m^2) Title 24 reference home, the air flow requirement was 73 cfm (34 L/s) and used 20.1 W.

Intermittent Exhaust: An intermittent exhaust system is modeled with the same requirements (air flow of 73 cfm (34 L/s) and 20.1 W power consumption) as the continuous exhaust, but being shut-off for 4 hours per day. These hours of non-operation vary seasonally to avoid peak cooling (in summer) and heating (in winter) times of day.

Heat Recovery Ventilator: Typical HRV installations do not operate continuously. In these simulations, they operated for half an hour then were off for half an hour. The HRV air flows in the HVI directory are typically much larger than the minimum 62.2 requirements. We chose one of the lowest air flow units in the HVI directory with an air flow of 130 cfm that used 124 W. This is about 35% more flow than simply doubling the 62.2 minimum requirement of 48 cfm for this house that would be required for its 50% duty cycle. We assumed that the supply and exhaust fans in the HRV consumed the same amount of power, i.e., 62 W. The heat added to the house by HRV operation was 55 W based on the 7W of required air power (derived from the product of air flow at a specific rated pressure). The temperature of air delivered by the HRV was determined from its rated apparent sensible effectiveness of 70%. The HRV is stand-alone and does not use the furnace blower or associated duct system to distribute the air in the house. The HRV was simulated for the two coldest climates: zones 1 and 16.

Continuous Exhaust plus Air Distribution: This is a continuous exhaust system (see above) which is augmented with a central fan integrated (CFI) supply that uses the furnace blower to intentionally draw outdoor air through a duct into the return and distribute it throughout the house using the heating/cooling supply ducts. The outdoor air duct is only open to outdoors during furnace blower operation and has a damper that closes when the furnace blower is off. This damper was assumed to have zero leakage when closed. The use of the HVAC system ducts means that air leakage and thermal losses are included in the operation of this ventilation system. In addition, the power consumption of the furnace fan is used in the energy use calculations. The furnace fan is a standard PSC blower with typical California ducts.

Continuous Supply (with dedicated Air Distribution): The continuous supply system will use a fan to supply filtered air from outside that then distributes the air throughout the house without using the furnace blower or the forced air heating and cooling ducts. Therefore the continuous supply air is not associated with any duct leakage or heat transfer effects. For continuous supply, the supply air is mixed with indoor air for tempering purposes. We will use a mixing ratio of 3:1 for indoor to supply air. The supply fan was sized to be four times the continuous exhaust requirements and provide 292 cfm (137 L/s) with a power consumption of 133 W, or which 14 W is air power and 119 W is heat added to the house.

Non-compliant systems

The systems below were simulated either because they provide good comparison cases or because they represent systems that are in current use. The systems below do not meet the proposed requirements for Title 24 or ASHRAE Standard 62.2.

Unvented House: This case represents the situation if there were no change to the Title 24 standards. It represents a house that is built without a mechanical ventilation system and did not have any significant window operation.

Vented House: This case represents the situation if people operated their windows as has been assumed since the 2001 version of Title 24. That is, that people will operated their windows a fixed amount when the ventilation rate drops too low. Such an approach may meet the intent of 62.2, but would not be compliant with the proposed Title 24 requirements. These simulations were for the unvented house, but with the minimum ventilation rate adder of 0.35 ACH used when air change rates fall below 0.35 ACH. This mimics the ventilation adder currently used in the Title 24 ACM.

Central Fan Integrated (CFI) Supply at 7% Fan Flow. This "7% solution" is a system that is in current use in various parts of the country. The central fan (i.e. the "blower") is on for 10 minutes, then off for 20 minutes. This system does not account for furnace blower operation for heating or cooling: the outdoor air supply duct was open and the blower was on for the first 10 minutes out of every 30 minutes regardless of the space conditioning system operating mode. Due to reduced operating time the net flows are therefore not 62.2 compliant. The outdoor air flow is 7% of the total blower flow. Because this is achieved by a fixed damper setting rather than damper modulation to achieve a fixed flow, this air flow is a fixed 7% of the furnace blower flow. I.e., 7% of heating fan flow during heating, 7% of cooling fan flow during cooling and 7% of cooling fan flow when ventilating only. A damper closes the outside air vent when the CFI is not operating (i.e. for 20 minutes out of every 30 minutes).

Central Fan Integrated Supply at 1/3 of the 62.2 Rate: This "1/3 solution" is a system that is in current use in various parts of the country. These simulations are the same as the 7% solution but with the air flow adjusted to be the 62.2 air flow rate rather than 7% of blower flow. Because the central fan operates one third of the time it provides one third of the 62.2 required air flow.

Source Control Ventilation

In addition to the specific technologies that meet 62.2, intermittent operation of kitchen and bathroom fans was included for source control. Intermittent bathroom fans operated for half an hour every morning from 7:30 a.m. to 8:00 a.m. These bathroom fans were sized to meet the 62.2 requirements for intermittent bathroom fans. From Table 5.1 in 62.2 this is 50 cfm (25 L/s) per bathroom. For houses with multiple bathrooms, the bathroom fans operated at the same time, so the 1,761 ft^2 house had a total of 100 cfm (50 L/s) and the 4,000 ft^2 house had a total of 150 cfm (75 L/s). Power requirements for these fans were 0.9 cfm/W based on recent California field survey data (Wilcox 2006), i.e. 55W for each 50 cfm fan.

Similarly, all simulations had kitchen fan operation. Based on input from ASHRAE Standard 62.2 members, the kitchen fans operated for one hour per day from 5 p.m. to 6 p.m. These kitchen fans were sized to meet the 62.2 requirements for intermittent kitchen fans. From Table 5.1 in 62.2 this was 100 cfm (50 L/s). Unfortunately, very few of the kitchen fans in the HVI directory had power consumption information. The smallest of those that do was selected for these simulations that had a flow rate of 160 cfm, and used 99W.

RESULTS AND DISCUSSION

Ventilation

Table 1 summarizes the mean annual air change rates for the three climates. The cold climate zone has greater natural infiltration and hence the greatest annual air change rate. The lowest ventilation rates are for the unvented house and the lowest 62.2 compliant rates are for intermittent exhaust that add 0.09 to 0.15 ACH to the annual average. The increase is larger for the climates with the low natural infiltration driving forces. The HRV and 0.35 ACH ventilation adder have the biggest effect – the adder more than doubles the ACH in Climate Zone 15.

Table 1 Mean Annual Air Changes Per Hour (ACH)										
	Simulation									
Climate Zone	1 Unvented House not 62.2 compliant	2 Cont. Ex.	3 Int. Ex.	4X HRV 50% ontime	5 CFI with Cont. Ex.	6 Supply	7 CFI 7%OA not 62.2 compliant	8 CFI 62.2 33% runtime not 62.2 compliant	Unvented House with 0.35 ACH Adder not 62.2 compliant	Cont. Ex. with 0.35 ACH Adder not 62.2 compliant
mild	0.24	0.37	0.35		0.44	0.46	0.30	0.31		
hot	0.24	0.42	0.39		0.51	0.45	0.45	0.35	0.52	0.59
cold	0.32	0.43	0.41	0.61	0.53	0.55	0.47	0.41	0.54	0.56

As well as average rates, it is important to look at incidence of some specific ventilation levels. For example, setting an absolute minimum hourly ventilation rate of 0.1 ACH can be considered desirable from a pollutant concentration perspective. Conversely, high ventilation rates would tend to have an excess energy penalty associated with them and limiting the upper rate to 0.5 ACH would limit the energy penalty. The number of hours that the average ventilation rate was below 0.1 ACH and above 0.5 ACH are shown in Table 2. The unvented house was the only case with a significant number of hours of under-ventilation and the warmer the climate, the greater the number of under-ventilated hours. The under-ventilation hours tend to occur in groups that are 3 to 5 hours in duration. The impacts of under-ventilation for time periods of this length depend on the specific pollutants of interest. The only other case to have any hours below 0.1 ACH was the intermittent exhaust. When the intermittent exhaust was off, the house is the same as the unvented house and ventilation rates can be low. The number of occurrences of high ventilation increased as temperature differences increased from the mild, to cold climate. Of particular note is the 0.35 ACH adder that results in most of the year being at high ventilation rates. It is the extra hours compared to the unvented house at the high air change rates that contributes to the extra energy required to condition the homes.

Table 2 Number of hours at Low and High Change Rates (8760 = one year)										
	Simulation									
Climate Zone	1 Unvented House *not 62.2 compliant*	2 Cont. Ex.	3 Int. Ex.	4X HRV 50% ontime	5 CFI with Cont. Ex.	6 Supply	7 CFI 7%OA *not 62.2 compliant*	8 CFI 62.2 33% runtime *not 62.2 compliant*	Unvented House with 0.35 ACH Adder *not 62.2 compliant*	Cont. Ex. with 0.35 ACH Adder *not 62.2 compliant*
	Below 0.1 ACH									
mild	429	0	13	-	0	0	0	0	-	-
hot	851	0	16	-	0	0	0	0	0	0
cold	238	0	25	0	0	0	0	0	0	0
	Above 0.5 ACH									
mild	452	768	736	-	1436	2466	598	601	-	-
hot	552	1755	1300	-	3319	2288	2154	864	5279	6730
cold	779	1939	1692	6958	4619	5659	3570	2207	6074	5796

Energy Use and Cost

Because the details of design features like the building envelope are common, we have elected to show the results relative to a base case, rather than as total. The advantage of this approach is that the common factors are subtracted out and the results reflect the difference due to the ventilation treatment chosen. We have elected to use the *unvented house* as the base case, but it is important to remember that that case is a non-compliant one. The results are shown in stacked bar charts where we have broken up the extra energy into a piece that represents the energy to induce the desired ventilation (Ventilation), the energy to distribute air (Distribution) and the energy to condition the air (Space Conditioning that combines heating and cooling). Distribution can provide benefits independent from ventilation and is shown as a separable building energy service. The energy use data was converted to dollars ($) assuming typical California utility rates of $0.13/kWh for electricity and $1.75/Therm (equivalent to $0.06/kWh) for gas.

Figures 1 and 1a show the results for the temperate climate. The total energy use for the reference house was 14,000 kWh (cost was $875) and was primarily heating. The exhaust systems have the least energy increase of about 1000 kWh (8% of the total space conditioning) and the intermittent system saves 200 kWh ($15) relative to the continuous system. The extra energy used by the Continuous Exhaust + CFI system and Continuous Supply systems is primarily due to the extra cost of distribution. The non-62.2 compliant systems used about 150 kWh less than the continuous exhaust but about 50 kWh more than the intermittent exhaust (the annual costs were about $40 to $50 more due to increased electricity use). Again this is due to the cost of distribution.

Figures 2 and 2a show the results for the hot dry climate. The total energy use for the reference house was 9,000 kWh (cost was $855) with about 20% more heating than cooling. The exhaust systems energy increase is very similar to the temperate climate at 930 kWh (10% of total space conditioning) and the intermittent system saves 300 kWh ($30) compared to the continuous system. The extra energy use of the Continuous Exhaust + CFI system is more than three times the exhaust only system mostly due to distribution. The distribution energy use was more significant for this climate because the cooling system capacity, air flow rate and therefore the distribution fan energy consumption were more than three times that of the temperate climate. The Continuous Supply system used about twice the energy of the exhaust only systems. The two CFI non-62.2 compliant systems used more then three times the energy of the exhaust only systems – primarily due to the cost of distribution. Lastly, the 0.35 ACH ventilation adder used in Title 24 ACM calculations uses more energy than the exhaust only systems because of higher uncontrolled ventilation rates in cooler weather.

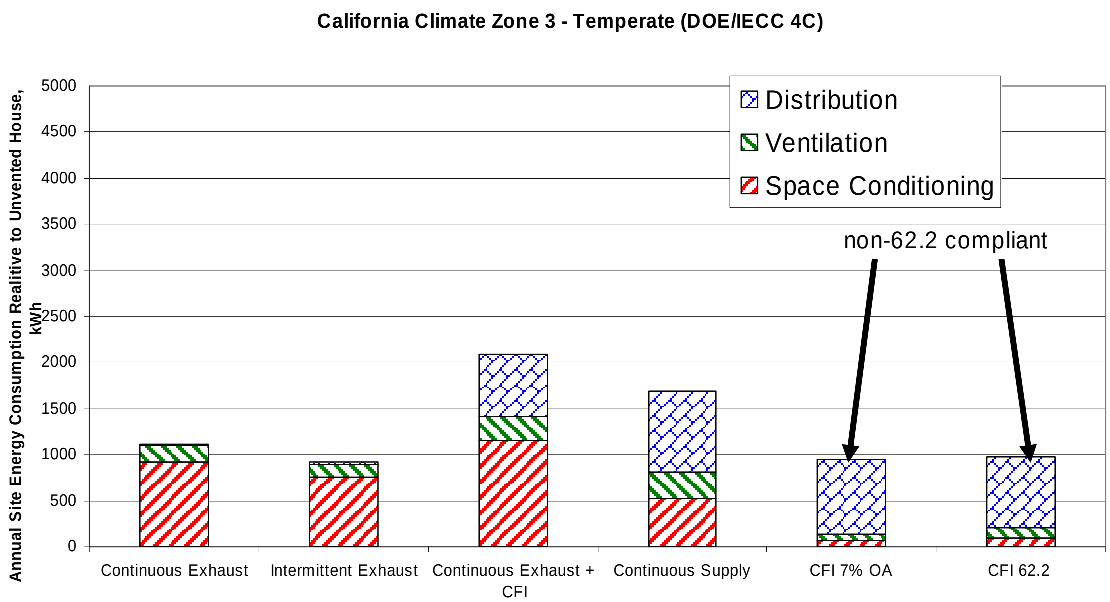

Figure 1. Energy use of ventilation relative to an Unvented House in a temperate climate.

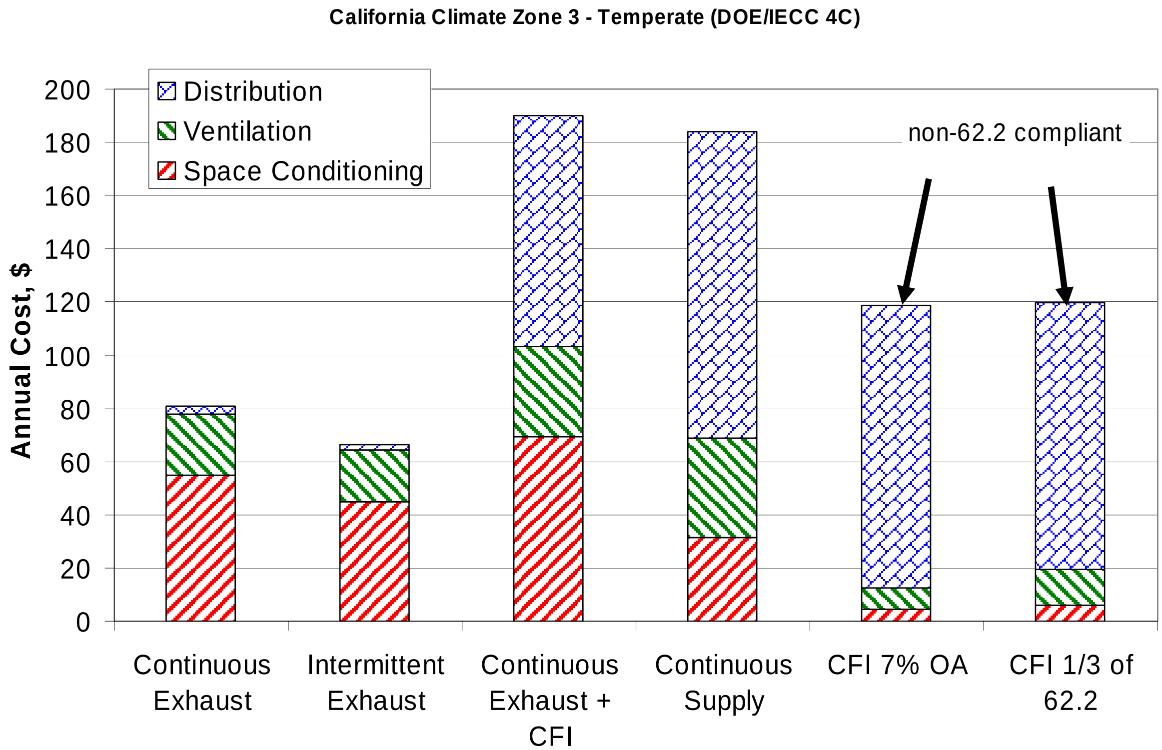

Figure 1a. Cost of ventilation relative to an Unvented House in a temperate climate.

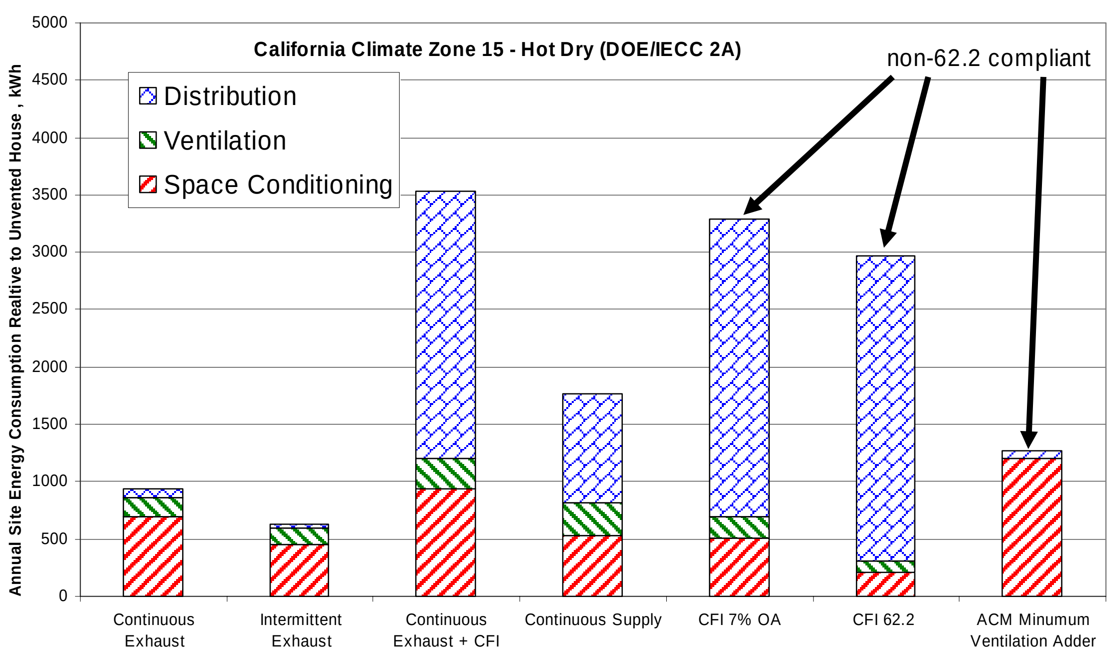

Figure 2. Energy Use of ventilation relative to an Unvented House in a hot dry climate.

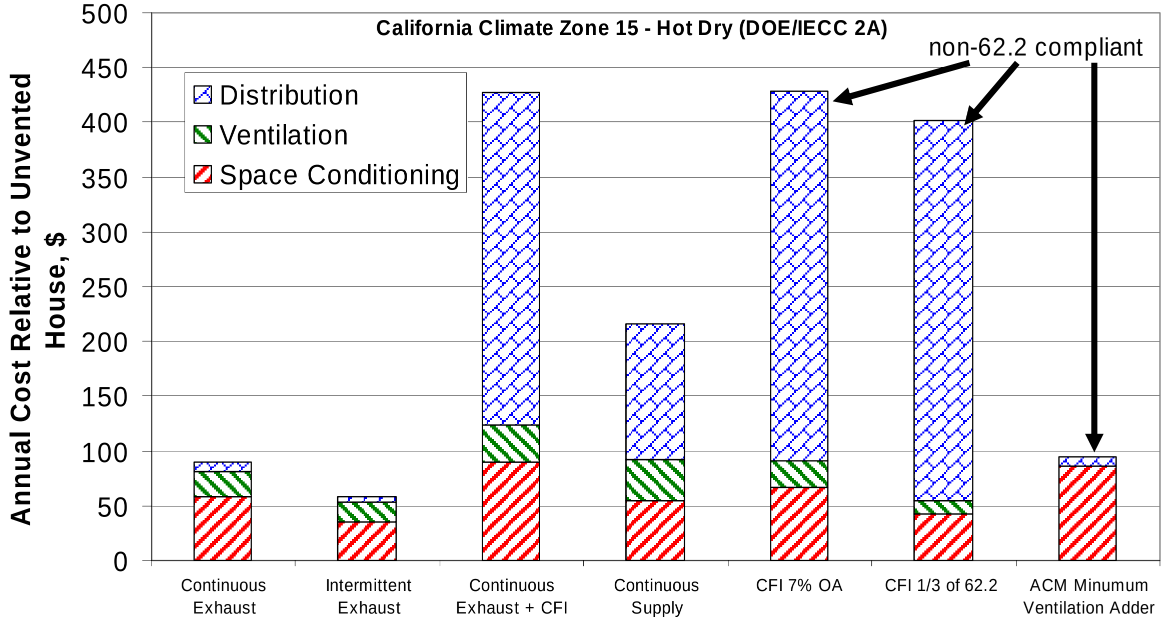

Figure 2a. Cost of ventilation relative to an Unvented House in a hot dry climate.

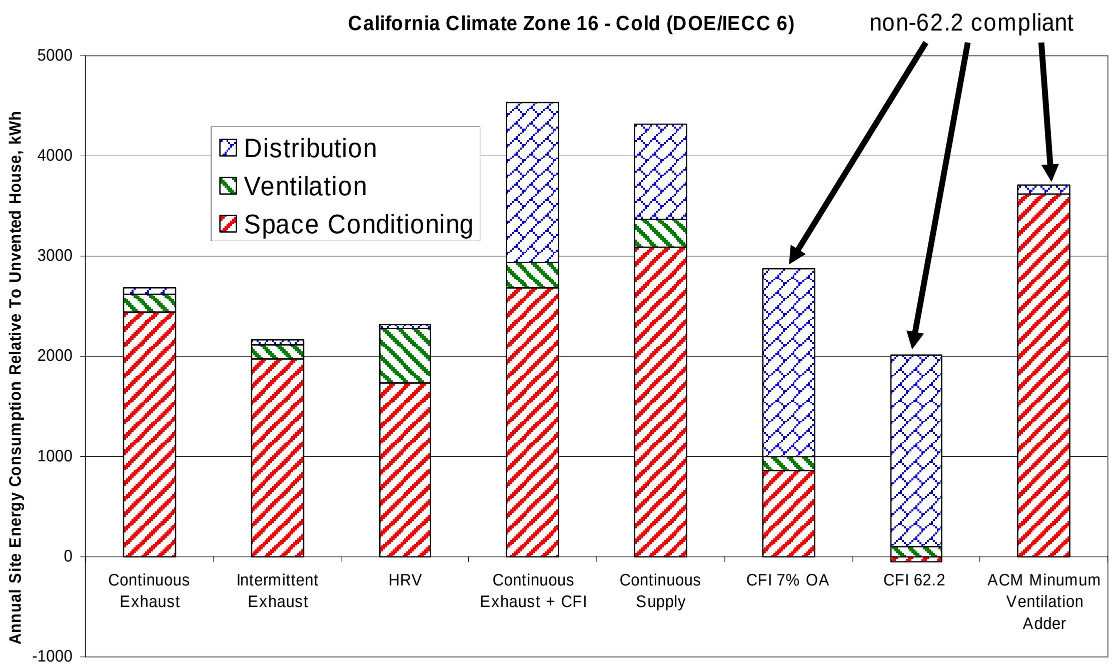

Figure 3. Energy use of ventilation relative to an Unvented House in a cold climate.

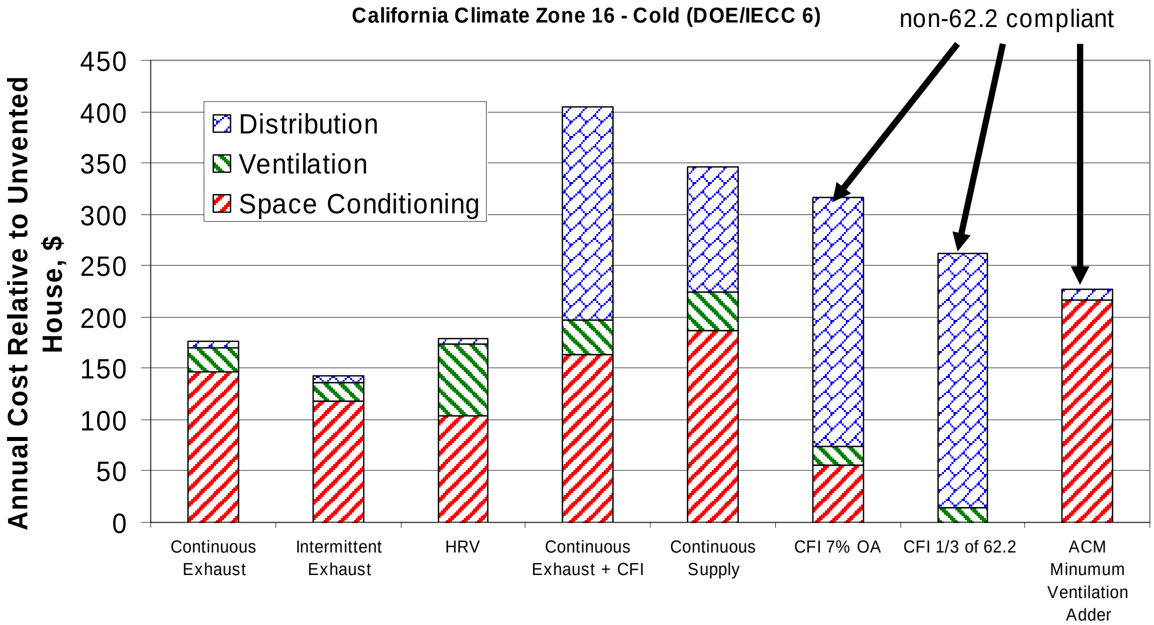

Figure 3a. Cost of ventilation relative to an Unvented House in a cold climate.

Figures 3 and 3a show the results for the cold climate. The energy used by the reference unvented house was 33,000 kWh ($2060). This is more than the previous two climates due to having a harsher climate with a lot greater heating requirement. The exhaust systems energy use increase is more than double that of the other climates at 2700 kWh (or 8% of total space conditioning and a costs of $150). The intermittent system saves about 500 kWh ($35) compared to the continuous system. The larger building load reduces the relative impact of operating the fan for distribution purposes to less than double the energy use of the continuous exhaust. The two CFI non-62.2 compliant systems distribution energy use offset the space conditioning savings. The CFI using the 62.2 outside air rate actually uses less space conditioning energy. This is because the waste heat from the distribution fan is electrically heating the building and this reduced the natural gas consumption. Because electricity is more expensive than gas, however, the space conditioning cost savings are negligible (less than $1). The 0.35 ACH ventilation adder used more energy than the exhaust only systems because of the space conditioning costs associated with the bigger temperature differences between inside and outside in this cold climate. The results for the HRV show that the lower space conditioning costs are offset by extra ventilation fan power.

Exhaust systems were found to be more energy efficient for several reasons. One of those reasons was that requirements of 62.2 do not account for the differences in combining of infiltration with balanced and unbalanced mechanical ventilation. Due to the non-linear nature of the pressure-flow relationship for building envelope leaks the infiltration and mechanical ventilation cannot be simply added. The effect of mechanical ventilation is always less than the simple addition of the infiltration and mechanical ventilation. For more details see ASHRAE Standard 136 (ASHRAE 2004) , Wilson and Walker (1990), Palmiter and Bond (1991) and ASHRAE Fundamentals, Ch. 27 (2005). The exception is for balanced systems (such as from an air to air heat exchanger) where the mechanical ventilation does not change the build envelope air flows and therefore the two flows can be added. This made the balanced systems have a larger impact on the total air exchange than does a supply system. Since the exhaust system had a lower impact on the total air exchange, its space conditioning impact was smaller and it used the least energy (and had the least operating cost) by virtue of producing fewer air changes.

For the systems that distribute and supply air there is a need to temper the incoming air to prevent cold drafts for occupants. For the supply system the tempering is achieved by mixing the incoming air with air form the house in the ratio of 3:1. This results in the lowest delivered ventilation air temperatures of all the systems studied. The critical times are in winter when it is cold outside. In our mild climate the lowest delivered air temperature is 54 °F (12 °C) and in our cold climate this drops to 48 °F (9°C). Because the

CFI systems mix incoming air with more house air (typically in ratios of about 15:1) the delivered air temperatures are less extreme at 57 °F (14 °C) and 54 °F (12 °C) respectively. These temperatures are likely to be too low for comfort so careful placement of supply registers and selection of grilles to minimize direct impingement on occupants would be recommended. The HRV did not mix incoming air with air from the house and all the tempering was provided by the heat exchange within the HRV. This led to lower air delivery temperatures than the supply and CFI cases – with a low of in 46 °F (8 °C) in the cold climate. HRVs can be installed so as to use the central distribution system and this would mitigate these low air delivery temperatures at the expense of additional fan power and duct leakage induced increased ventilation.

Air Distribution

All three climates show that providing active air distribution can use substantially more energy. Air distribution usually increases fan power and also induces thermal losses depending on duct performance. From a strictly energy performance standpoint all ventilation strategies that require air distribution are less desirable. Air distribution, however, can provide additional services such as distributing conditioning or outdoor air to reduce variations between zones. The results show that the lowest operating cost for distributing supply air is to use a dedicated supply system. This is due to the reduced power requirements of the dedicated supply fans compared to using the furnace or air conditioner blower.

The costs and benefits, therefore, of ventilation and air distribution systems must be evaluated independently. This, of course, can only be done in a systems approach in which all key requirements are being met. For example, any supply system that does not precondition the supply air will require air distribution to temper it to an acceptable level.

Natural Ventilation

The current version of California's Energy Code has an implicit assumption that occupants will naturally ventilate in a specific way, when infiltration alone is insufficient to meet minimum requirements. We called this the minimum ventilation adder. In the climates where we have compared them, the mechanical methods that did not require dedicated distribution saved energy compared to ventilation adder approach.

SUMMARY, CONCLUSIONS AND RECOMMENDATIONS

In this paper we used a simulation approach to determine the likely energy impacts of specific residential ventilation requirements and strategies for three different climates. The energy used to provide ASHRAE 62.2 complaint ventilation was about (the cost was in the range of $50-150/yr) for the most efficient options, which typically represented less than 10% of the heating and cooling costs.

In every climate the intermittent exhaust system proved to be the most energy efficient system for meeting the proposed requirements—followed by the continuous exhaust system. These systems were even more energy efficient than the non-compliant systems.

Exhaust systems were found to be more energy efficient for several reasons. One of those reasons was that requirements of 62.2 do not account for the addition of infiltration and mechanical ventilation. Since the exhaust system will have a lower impact on the total air exchange, its space conditioning impact will be smaller and it will be the more cost-effective approach, but by virtue of producing fewer air changes. The issue of the role of infiltration and superposition and their application to ventilation standards requires more research.

Another conclusion of our analysis is that central fan integrated systems use substantially more energy than other systems. This energy is principally required because of the enhanced air distribution provided by these systems. While neither Title 24 nor ASHRAE 62.2 require it, air distribution can presumably have non-energy benefits in the form of increased comfort and improved indoor air quality.

Air distribution systems are used to distribute heating and cooling, to distribute fresh air, and to filter, clean and recirculate indoor air. As such they are a separable building function from the rest of the HVAC system and should be treated separately both in regulation and design. One advantage of treating distribution as a separate function is that it will allow efficiency advances in distribution to be made independently from, for example, cooling systems. Further research is needed to determine the value of air distribution to reducing sizing of the rest of the HVAC system.

If distributing of outdoor air is a key criteria, our results indicate that the best way to do that is with a dedicated supply ventilation system. If redistributing air for comfort or filtration means is important then one of the compliant systems with central-fan integration should be selected. Not considered in this study, however, is the use of high-efficient variable speed motors that could be used with the central fan to distribute low volumes of air energy efficiently. Such evaluations might suggest valuable alternative systems.

Heat Recovery Ventilators have the least cost of conditioning air in extreme climates but inclusion of the HRV fan energy cost makes the total cost about the same as for an exhaust only system. The HRVs modeled here did not, however, have any air distribution associated with them and so at times low supply air temperatures might cause discomfort depending on the design. Central-fan integrated HRV designs are also in use. Although not modeled in this study, we would expect such a design to have an air distribution cost in line with the other central-fan integrated systems. As HRV/ERV systems become more common, evaluations of these options would be appropriate.

Non-mechanically ventilated homes have several hundred hours of the year when ventilation rates fall below 0.1 ACH. This is of concern from an indoor air quality point of view.

While not the focus of this study, we determined that the following research needs exist:

- A broad based field study to determine envelope and duct air leakage of current new construction and how commonly used mechanical ventilation systems perform with such leakage.
- A study to determine how contaminants are transported from garages and other buffer zones and to determine if Carbon Monoxide alarms are necessary.
- Evaluation of the need to have air distribution systems in order to provide acceptable indoor air quality.
- Development and testing that would allow additional ventilation credit to be given to economizers, direct evaporative coolers and other systems that provide ventilative cooling.

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